INTRODUCTION

The response of a room may result in a high sound pressure level at certain frequencies which coincide with the natural frequencies of the room, even with small amount of sound energy [1]. This leads to an uneven spatial distribution of acoustic energy in the room, namely the nodes and antinodes of sound pressure. Consequently, low-frequency resonances can have a significant impact on the sound quality of an audio system for example, and thus contribute to prejudicial audible effects in the sound rendering in the listening environment, such as showrooms or recording studios [2].

Usual soundproofing materials are generally cheap and simple to implement to address noise issues, but not always effective in the low-frequency range [3]. Alternatively active noise control methods can be used and are able to counteract the noise selectivity on the principle of destructive interferences [4]. Good performances can be observed in case of simple geometries or with stationary tones. In three-dimensional sound field where noise is difficult to predict, the number of required secondary sound sources quickly becomes prohibitive, and the distributed control algorithms may be complicated for implementing.

The use of electroacoustics resonators can assist for absorbing the acoustic energy propagating near the diaphragms of loudspeakers. With the help of specific electrical load connected to the electrical terminals, the transducer dynamics are altered and the sound absorption capability is then improved by extending the control bandwidth around the transducer resonance. In a recent study, the impedance-based control has been implemented in a room for the application of semi-active modal equalization [5]. From different spatial arrangements of electroacoustic resonators located in the corners where antinodes are particularly pronounced, significant damping of the low-frequency resonances has been provided. As the choice of the location and orientation of electroacoustic resonators is quit difficult, we want to investigate the optimization of the spatial arrangement in a room.

This article introduces the concept of electroacoustic resonator from the characteristics of the closed-box loudspeaker. The intake of coupling a specific electrical load across the transducer terminals is discussed in terms of sound absorption capability. For the semi-active modal equalization, the location of the electroacoustic resonators are justified with the help of the identification of the eigenfrequencies of a specific room. A methodology based on an experimental design is proposed for the optimization of the spatial arrangement.

ELECTROACOUSTIC RESONATORS

Governing equations

Let us consider an electrodynamic loudspeaker in a closed box as a lumped parameter electromechanical system in order to illustrate the basic theory 1 [6]. For small displacements and below the first modal frequency of the diaphragm, the governing equation of the mechanical part follows from the Newton's second law and can be expressed using phasor representation as

$$S\underline{p} = \left(j\omega M_{ms} + R_{ms} + \frac{1}{j\omega C_{mc}}\right)\underline{v} - Bl\,\underline{i} \tag{1}$$

where $\underline{p}$ is the driving pressure acting on the transducer diaphragm (in Pa), $\underline{v}$ is the diaphragm velocity (in m s^{-1}), $\underline{i}$ the electrical current flowing through the voice coil (in A). For the model parameters, S is the effective piston area (in m^2), Bl is the force factor (in N A^{-1} ; product of B the magnetic field amplitude and l the length of the wire in the voice coil), M_{ms} and R_{ms} are the mass (in kg) and mechanical resistance (in N s m^{-1}) of the moving body.

Here, $C_{mc} = (1/C_{ms} + \rho c^2/V_b)^{-1}$ is the equivalent mechanical compliance (in N m^{-1}) accounting for both the flexible edge suspension and spider of the loudspeaker C_{ms} and the enclosure, where

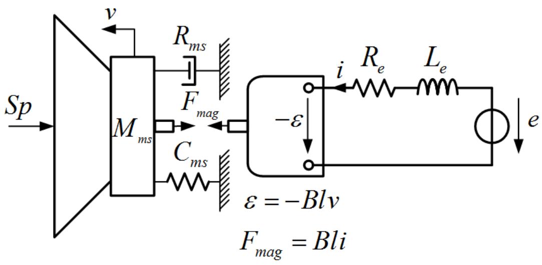

FIGURE 1: Schematic representation of a loudspeaker with terms of electromechanical coupling induced

ρ and c are the density (in $\mathrm{kg\,m^{-1}}$) and celerity of air (in $\mathrm{m\,s^{-1}}$) and V_b is the volume of the enclosure (in $\mathrm{m^3}$).

The governing equation of the electrical side is based on Kirchhoff's laws and can be written as

$$\underline{e} = (j\omega L_e + R_e)\,\underline{i} + Bl\,\underline{v} \tag{2}$$

where $\underline{e}$ is the input voltage applied across the electrical terminals (in V), L_e and R_e are the self inductance of the voice coil (in H) and the dc resistance (in Ω).

These two equations contain terms of electromechanical coupling that arise from the interaction of the mechanical and electrical variables. The coupling term $Bl\,\underline{i}$ represents the Laplace force $\underline{F}_{mag}$ induced by the current circulating through the coil (in N) and $-Bl\,\underline{v}$ is the back electromotive force $\underline{\varepsilon}$ induced by the motion of the coil within the magnetic field (in V). The radiation impedance is excluded of the following development with a view of providing general properties of the loudspeaker (apart from the radiation conditions of the diaphragm).

Connecting an external electrical load

By connecting an electric load of complex impedance $\underline{Z}_L$ at the transducer terminals, the voltage $\underline{e}$ applied across the transducer terminals becomes

$$\underline{e} = -\underline{Z}_L\,\underline{i} \tag{3}$$

and the electrical current $\underline{i}$ flowing through the coil can be written as

$$\underline{i} = \frac{1}{\underline{Z}_e + \underline{Z}_L}\underline{\varepsilon} \tag{4}$$

where $\underline{Z}_e = R_e + j\omega L_e$ is the blocked electrical impedance of the voice coil. When designed properly, the shunt electrical impedance $\underline{Z}_L$ can make a functional relationship between the induced voltage $\underline{\varepsilon}$ and electrical current $\underline{i}$, thus taking precedence over the transducer dynamics. The methodology for designing the specific electrical load of complex impedance $\underline{Z}_L$ is detailed in [7].

Acoustic absorption capability

A closed form expression of the specific acoustic admittance at the transducer diaphragm can always be derived after eqs. (1-3) regardless of the load connected across its terminals. Normalizing relative to the characteristic impedance of the medium ρc, the specific acoustic admittance ratio can be expressed as

$$\underline{y} = \rho c \frac{\underline{v}}{\underline{p}} \tag{5}$$

This dimensionless parameter reflects the motion (response) of the diaphragm that is caused by the driving acoustic pressure. By combining Eqs. (1-5), the generalized velocity response of the transducer diaphragm to any surrounding sound field can be expressed as

$$\underline{y} = \rho c S \frac{\underline{Z}_e + \underline{Z}_L}{(\underline{Z}_e + \underline{Z}_L)\,\underline{Z}_{mc} + (Bl)^2} \tag{6}$$

where $\underline{Z}_{mc} = j\omega M_{ms} + R_{ms} + 1/(j\omega C_{mc})$ is the mechanical impedance of the moving body of the closed-box loudspeaker. The corresponding sound absorption coefficient α at normal incidence can be derived as

$$\alpha = 1 - \left|\frac{1-\underline{y}}{1+\underline{y}}\right|^2 \tag{7}$$

Figure 2(a) illustrates the computed specific acoustic admittance $\underline{y}$ at the diaphragm when connecting a low-range Scan-Speak 30W/4558T00 loudspeaker (see. Table 1) mounted in sealed enclosure of volume $V_b = 45L$ to a specific shunt electrical load. As clearly shown with the corresponding sound absorption coefficient in Fig. 2(b), the acoustic impedance control bandwidth is effective in the low-frequency range.

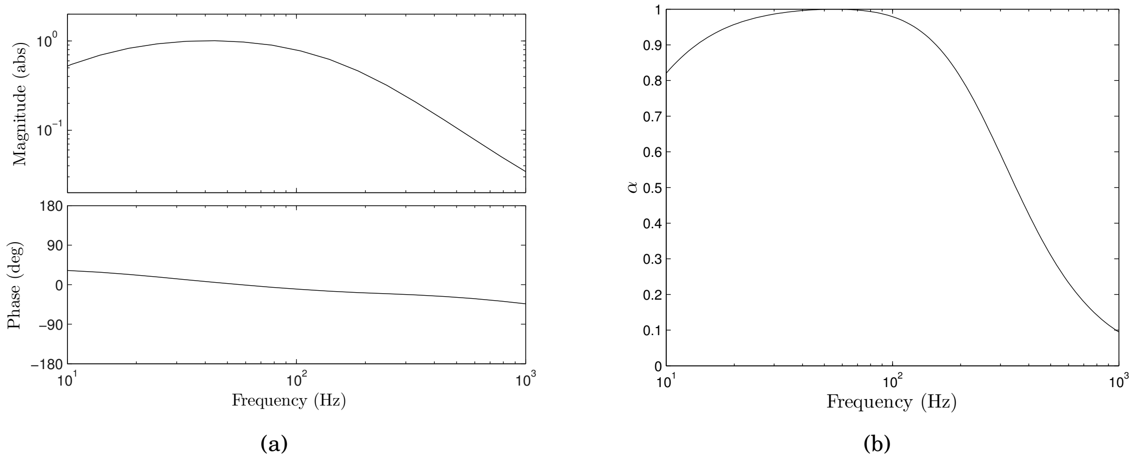

FIGURE 2: Computed specific acoustic admittance (left) and corresponding sound absorption coefficent (right).

TABLE 1: Small signal parameters of the Scan-Speak 30W/4558T00

Parameter	Notation	Value	Unit
dc resistance	R_e	2.6	Ω
Voice coil inductance	L_e	0.83	mH
Force factor	Bl	10.5	NA^{-1}
Moving mass	M_{ms}	135	g
Mechanical resistance	R_{ms}	2.88	$Nm^{-1}s$
Mechanical compliance	C_{ms}	0.65	mmN^{-1}
Effective area	S	466	cm^2
Resonance frequency	f_s	17	Hz

MODAL EQUALIZATION OF THE ROOM

The present work considers a room which is almost parallelepiped, used as showroom for audio system (cf Fig. 3). This is a hard-wall room with a truncated corner (length L_x = [min. 5.62 m, max. 7.02 m], width L_y = [min. 3.70 m, max. 5.10 m] and height L_z = 2.70 m). The total area is 130,2 m^2 and the volume is 94 m^3. From the studied geometry the low-frequency distribution of acoustic energy in enclosed spaces can be assessed, as well as the potential performances of electroacoustic resonators to damp the resulting resonances.

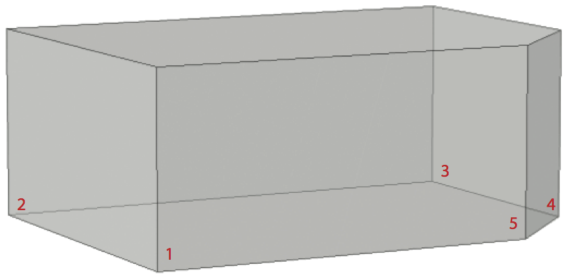

FIGURE 3: Geometry of the studied room

Eigenfrequencies identification

From a frequency point of view, the object of modal damping is to reduce the amplitudes of mode resonances, by presenting specific acoustic resistances at specific room resonance frequencies. With a view to achieving such room modal damping, the first stage is the identification of the room modes. The explicit evaluation of the eigenvalues and eigenfunctions of complex geometry room is quite difficult and cannot be derived after solving the eigenvalues from the wave theory of room acoustics [2]. For ease of calculation, the eigenfrequencies are identified with the help of applications of numerical methods such as Comsol Multiphysics (finite element method).

Figure 4 shows the computed sound pressure level in the room for a selection of eigenfrequencies between 68 Hz and 90 Hz for c = 343 m.s^{-1}, with the corresponding mode structure given by subscripts $f_{(n_x,n_y,n_z)}$. This graphical representation clearly illustrates the nodes (in blue) and antinodes of pressure (in red).

Placement of electroacoustic resonators

The analysis of the acoustic energy distribution shows that the electroacoustic resonators must be carefully placed in the room. At low frequencies, typically where the size of electroacoustic resonators becomes small relative to the wavelength, the coupling with the room is inefficient when they are located on pressure nodes [1]. For optimal performance, it is best to place them on pressure antinodes, while directing the transducers diaphragm according to the modes structure to be damped. The electrocousitc resonators also are more efficient when they are located in the corners with closed angles (bottom corners #1, #2 and #3 for the studied room), where the sound pressure energy is the most important.

In [5] the study showed that the magnitude of the low-frequency resonances of the tested room was greatly reduced, even with a very small equivalent absorption surface. The measured gains of the experimental assessment were largely related to the location of the resonators, as well as the orientation of the diaphragm.

The control of acoustic impedance loses its effectiveness when the acoustic waves which reach the loudspeaker diaphragms are not under normal incidence. Depending on the modes, an optimal location which is highlighted for damping a mode may not correspond to the optimal placement for another modes.

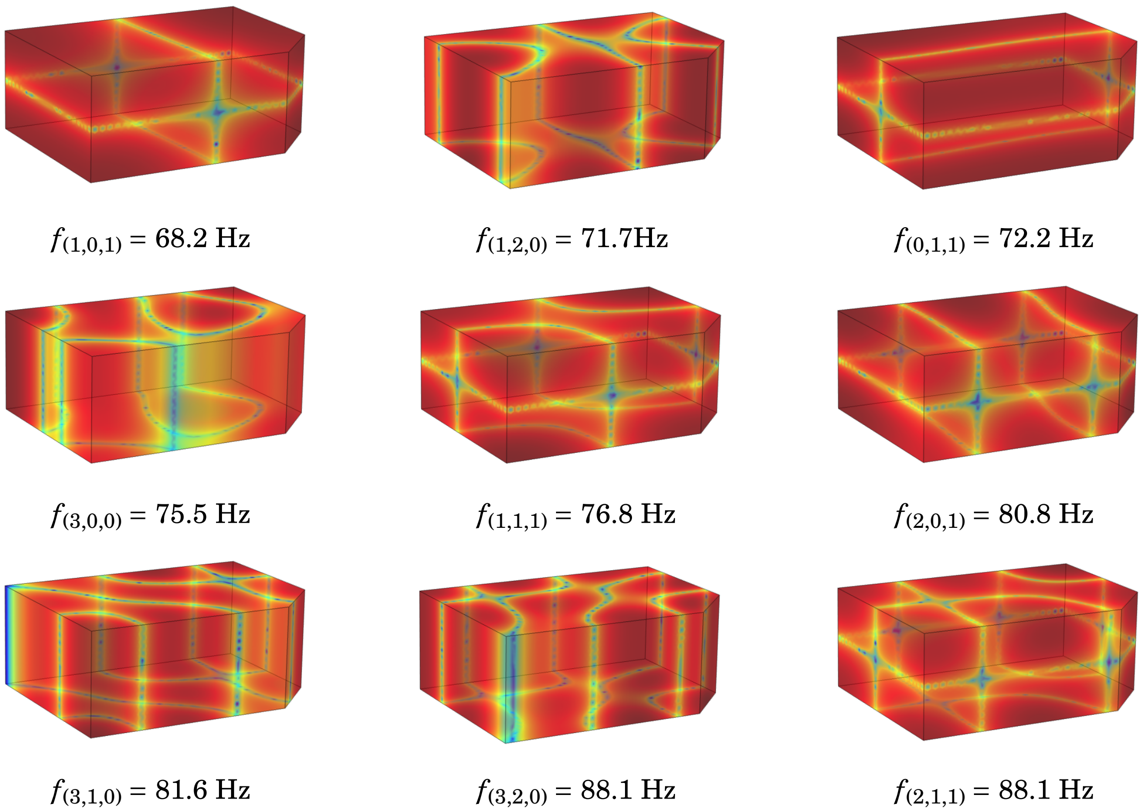

FIGURE 4: Computed sound pressure level illustrating a selection of modes of the room between 68 Hz and 90 Hz.

OPTIMIZATION METHODOLOGY

Strategy

The optimization of electroacoustic resonators is very interesting for semi-active room equalization. For a small number of resonators, the choice of their location and orientation in the room is quit difficult due to the large number of resonances in the acoustic impedance control bandwidth. Depending of the spatial arrangement of resonators in the room, the damping can be particularly effective for a specific mode, but may not be the best setting for contributing to the equalization in the low-frequency range.

The methodology of experimental design aims precisely to solve the strategic problem. The primary objective of this experimental design is to determine the best configuration of electroacoustic resonators for the semi-active room modal equalization in the low frequency range. The second objective is to analyze the significant effects and interactions of experimental factors, i.e the location of resonators in relation to their orientation, for further improvements of resonators application.

Due to the many possibilities of orientation, let us consider the 7 elementary cases where the loudspeaker diaphragm is facing edges along the x-axis, y-axis, z-axis, xy-axis, xz-axis, yz-axis and xyz-axis. For an experimental assessment with electroacoustic resonators placed in each of the 10 corners of the room, we must carry out 7^{10} experiments. Taking into account of the acoustic energy distribution, the choice of selecting only 3 resonators at fixed locations in the bottom corners #1, #2 and #3 for the study allows us to avoid the exponential expansion of the number of experiments with the number of factors. From the wave theory of room acoustics, we know that the first modes depend mainly on the largest dimension of the room (i.e the x-axis). We suppose that the configurations along the xz-axis and the yz-axis are less efficient than the xy-axis for the equalization of the first resonances. We do the choice of simplifying the model by selecting the 5 combinations x-axis, y-axis, z-axis, xy-axis and xyz-axis.

Response type

We need to choose the type of response for the experimental design, which has to be straightforward (i.e only one data per experiment) and representative of the semi-active room modal equalization efficacity. We choose as response type the average absolute deviation of the sound pressure level (SPL) data set, which corresponds to the average distance of the SPL data set from its SPL average L_{eq}. The SPL data are computed between 22.4 Hz and 178 Hz with a step of 0.1 Hz, in order to ease the numerical simulation time. The average absolute deviation of the set of n SPL data is expressed as

$$\frac{1}{n}\sum_{i=1}^{n}\left|10^{\frac{SPL_i}{20}} - 10^{\frac{L_{eq}}{20}}\right| \tag{8}$$

where the SPL average L_{eq} is computed from the values by third-octave bands.

Model establishment

For analyzing the significant effects and interactions of experimental factors, i.e observing the influence of the location of the electroacoustic resonators in the studied room, as well as their orientation, we define our own linear system of equation. Let us consider a linear model with first degree interactions expressed as

$$Y(x) = \mu + \sum_{i=1}^{N}\alpha_i x_i + \sum_{j=1}^{N}\beta_j x_j + \sum_{k=1}^{N}\gamma_k x_k + \sum_{i,j=1}^{N}\alpha_i\beta_j x_i x_j + \sum_{i,k=1}^{N}\alpha_i\gamma_k x_i x_k + \sum_{j,k=1}^{N}\beta_j\gamma_k x_j x_k \tag{9}$$

where the coefficients μ, α_i, β_i, γ_j, $\alpha_i\beta_j$, $\alpha_i\gamma_k$ and $\beta_j\gamma_k$ are the effects of the factors x. The factors x_i, x_j and x_k represent the resonators for the 3 locations in the bottom corners #1, #2 and #3, and $i, j, k = 1, ..., N$ represent the number of corresponding orientations (N = 5). It seems obvious that we can assign only one orientation to each of the resonators per experiment. The coefficient μ is the constant effect of the totality of experiments, the coefficients α_i, β_j and γ_k are the main effects, and the coefficients $\alpha_i\beta_j$, $\alpha_i\gamma_k$ and $\beta_j\gamma_k$ are the first order interaction effects. The coefficient Y represents the results of the experiments (m = 125) of the design for the 15 factors.

The model of eq. (9) may be written in matrix form as

$$Y = X\beta \tag{10}$$

where Y is a m x 1 vector of the experimental data, X is a m x p matrix of the model, β is a p x 1 matrix of the model coefficients, p being equal to 1 + number of locations x number of orientations x (1 + number of orientations). The coefficients of the model can be estimated through the least square resolution in order to determine the best configuration for the semi-active room modal equalization [8]. The unknown terms β can be obtained from the formula

$$\beta = (X'X)^{-1}X'Y \tag{11}$$

In the second part of analysis, we can interpret the significant effects and interactions of experimental factors by comparing the data with a statistical distribution such as the normal distribution.

CONCLUSION

In this paper we discussed a practical realization of electroacoustic resonators in view of controlling the low-frequency sound field in closed spaces. A simple engineering approach employing an arrangement of electrodynamic loudspeakers the terminals of which are connected to a specific electrical load has been presented. Through judicious control of acoustic impedance in a test room a significant damping of the dominant natural resonances can be achieved by absorbing incident sound waves. As the proposed optimization methodology is still in process because of the large number of experiments, further results will be presented at the meeting.

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